NACA

RESEARCH MEMORANDUM

ANALYTICAL INVESTIGATION OF DISTRIBUTION OF CENTRIFUGAL STRESSES AND THEIR RELATION TO LIMITING OPERATING TEMPERATURES IN GAS-TURBINE BLADES

By Richard H. Kemp, and William C. Morgan

Flight Propulsion Research Laboratory
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS
WASHINGTON
April 12, 1948

— File

National Aeronautics and
Space Administration

Langley Research Center
Hampton, Virginia
23665

NASA

JUN 1 6 1983

Reply to Attn of: 139A

TO: Distribution

FROM: 180A/Security Classification Officer

SUBJECT: Authority to Declassify NACA/NASA Documents Dated Prior to January 1, 1960

Effective this date, all material (informal correspondence) classified by this Center prior to January 1, 1960, is declassified. This action does not include material derivatively classified at the Center upon instructions from other Agencies.

Immediate re-marking is not required; however, until material is re-marked by lining through the classification and annotating with the following statement, it must continue to be protected as if classified:

> "Declassified by authority of LaRC Security Classification Officer (SCO) letter dated June 16, 1983," and the signature of person performing the re-marking.

If re-marking a large amount of material is desirable, but unduly burdensome, custodians may follow the instructions contained in NHB 1640.4, subpart F, section 1203.604, paragraph (h).

This declassification action complements earlier actions by the National Archives and Records Service (NARS) and by the NASA Security Classification Officer (SCO). In Declassification Review Program 807008, NARS declassified the Center's "Research Authorization" files, which contain reports, Research Authorizations, correspondence, photographs, and other documentation. Earlier, in a 1971 letter, the NASA SCO declassified all NACA/NASA formal series documents with the exception of the following reports, which must remain classified:

Document No.	First Author
E-51A30	Nagey
E-53G20	Francisco
E-53G21	Johnson
E-53K18	Spooner
SL-54J21a	Westphal
E-55C16	Fox
E-56H23a	Himmel

JUN 2 3 1983

If you have any questions concerning this matter, please call Mr. William L. Simkins at extension 3281.

Jess G. Ross
2898

Distribution:
SDL 031

cc:
NASA Scientific and Technical
Information Facility
P.O. Box 8757
BWI Airport, MD 21240

NASA--NIS-5/Security
180A/RIAD
139A/TU&AO
6-15-83
139A/WLSimkins:elf 06/15/83 (3281)
139A/JS 6-15-83

31-01 HEADS OF ORGANIZATIONS
HESS, JANE S.,
MAIL STOP 188 BLDG 1194

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

ANALYTICAL INVESTIGATION OF DISTRIBUTION OF CENTRIFUGAL STRESSES AND THEIR RELATION TO LIMITING OPERATING TEMPERATURES IN GAS-TURBINE BLADES

By Richard H. Kemp, and William C. Morgan

SUMMARY

An analytical investigation was made to determine the centrifugal-stress distributions in six turbine-blade designs representative of current turbine blades. The investigation also included some consideration of temperature effects in relation to centrifugal stress.

Sufficient experimental data were available on one type of turbine blade to permit investigation of the effect of service operating temperatures on the ability of the turbine blade to withstand 100-hour operation at several different speeds. The stress analysis for this blade indicated that failure would occur at a location approximately halfway between the base and the tip of the blade. The number of service failures that have occurred in this region corroborates the analytical indication of critically low material strength.

INTRODUCTION

A program is in progress at the NACA Cleveland laboratory to investigate the problems associated with the stresses and the thermal conditions that occur in various types of gas-turbine blade employed in turbojet engines and turbine-propeller units. As a part of this general program, calculations were made to determine the stress distributions that exist in six current turbine blades when these blades are subjected to the centrifugal forces present during full-speed service operation.

Superior design could be achieved if additional information were available on the stress conditions that exist in operating turbine blades. Analyses that provide a basis for comparing the centrifugal-stress distributions inherent in jet-engine turbine blades of several designs currently in use were therefore made and are presented. Comparison was made by considering the stress distributions in the individual blades in terms of percentage of blade length. The turbine blades that were selected for this survey differed considerably in physical dimensions, but the design characteristics were basically similar except for the integrally shrouded blade.

The high temperatures that exist in operating turbine blades effect a considerable reduction in the strength of the material. Experimental data on the stress-rupture characteristics of high-temperature materials were readily available. In order to calculate the effect of temperature on the strength of the material in an individual blade, however, it is necessary to know the operating temperature at a number of points on the blade for a given set of conditions. Unpublished experimental temperature data were available for one type of turbine blade studied in this investigation. These experimental data were correlated with stress-rupture data and with the calculated distribution of centrifugal stress; this correlation permitted comparison between rupture stresses and centrifugal stresses for several service-operation turbine speeds.

A complete analysis of the operating stresses would include consideration of the effects of factors other than centrifugal stress and stress-rupture limitation of the material as affected by operating temperature. Twist, creep, bending, vibratory fatigue, gas erosion, aerodynamic flutter, and thermal stresses caused by uneven temperature distribution should be considered in a comprehensive investigation. The investigation reported serves only as an initial approach to the problem of failure in turbine blades.

BLADES

The blade designs selected for the analysis are designated A, B, C, D, E, and F. Three-quarter view sketches of these blade designs are presented in figure 1. Turbine-blade designs A and B are used in turbines driving centrifugal compressors; blades C, D, E, and F are used in axial-flow compressor units. Blade C represents the type used in a shrouded turbine wheel. The turbine blades of designs E and F are used in the first and second stages, respectively, of a two-stage turbine. These six blades are representative of current American gas-turbine-blade design.

SYMBOLS

The following symbols are used in this analysis:

A_X cross-sectional area at distance X from tip, (sq in.)

A_x cross-sectional area at distance x from tip, (sq in.)

A_Y area beneath curve $A_x\,(R + L - x)$ from cross section A_X to tip of blade, (sq in.)

F centrifugal force, (lb)

F_X centrifugal force at distance X from tip, (lb)

g acceleration of gravity, (in./sec^2)

K constant dependent on density and speed of rotation

L length of turbine blade, (in.)

m mass, (lb-sec^2/in.)

R radius of turbine rotor, (in.)

r radius of revolution for mass, (in.)

X distance from tip to cross section where stress is to be determined, (in.)

x distance from tip to any cross section, (in.)

ρ density, (lb/cu in.)

σ_X stress at cross section A_X, (lb/sq in.)

ω angular velocity, (radians/sec)

ANALYTICAL PROCEDURE

The first part of the analysis was concerned with distribution of centrifugal stress along the axial lengths of the individual blades at full rated turbine speeds. Comparison was then made by considering the stress distribution in terms of percentage of blade length. The second part of the analysis consisted of an examination of the relation between centrifugal-stress distribution in one of the turbine blades and the rupture stress as limited by experimentally determined temperature distribution.

Distribution of centrifugal stress. - Centrifugal force on any object revolving about a fixed axis can be determined from the general formula

$$F = m\omega^2 r \tag{1}$$

A conventionalized sketch of a turbine blade with the dimensional notations used in the derivation is presented in figure 2. By substituting $(R + L - x)$ for r in equation (1), and $A_x \frac{\rho}{g}\, dx$ for dm, the following equation may be written to represent the centrifugal force acting on the incremental volume, $A_x dx$:

$$dF_X = A_x \frac{\rho}{g}\ \omega^2\ (R + L - x)\ dx \tag{2}$$

When integrated, equation (2) becomes

$$F_X = \frac{\rho\,\omega^2}{g} \int_0^X A_x\ (R + L - x)\ dx \tag{3}$$

Then the stress at the cross section A_X is expressed by the following equation:

$$\sigma_X = \frac{\rho\,\omega^2}{A_X\, g} \int_0^X A_x\ (R + L - x)\ dx \tag{4}$$

In order to provide a simple and quick determination, the solution of equation (4) was made in part by graphical methods. The cross-sectional area of each of the turbine blades at various locations was obtained. From this information, a curve of cross-sectional area against turbine-blade length was plotted for each individual blade. The value A_x for any blade could then be read from the proper curve. The constants R and L were known and by arbitrary selection of the variable x sufficient points were established to obtain a curve of $A_x\ (R + L - x)$ for each turbine blade.

865

The centrifugal stress was then calculated from a simplified equation derived from equation (4).

$$\sigma_X = K \frac{A_Y}{A_X} \tag{5}$$

Solution of equation (5) was made for a sufficient number of values of A_X to establish a curve of stress distribution for each blade.

Effect of temperature distribution. - Experimental data were available on the temperature distribution in turbine blade of type B during engine operation. The axial temperature distributions of this blade for turbine speeds of 10,000, 11,000, and 11,500 rpm are shown in figure 3. These data were obtained from an unreported investigation made at the NACA Cleveland laboratory.

In order to determine the maximum allowable stresses in the material of turbine blade B for these service conditions, stress-rupture curves for the blade material (Vitallium) were used. The 100- and 1000-hour curves (data obtained from reference 1) shown in figure 4 were selected in order to permit estimation of the service life of the turbine blade. Correlation of the information provided by the curves of figures 3 and 4 permitted determination of the theoretical maximum allowable stress-distribution curves for turbine blade B at three operating speeds and two conditions of time.

RESULTS

The axial distributions of centrifugal stress and of cross-sectional area of the six types of turbine blade under full-speed operation of the turbines in which they are used are presented in figures 5 to 10. In figure 11 the data representing the centrifugal-stress distributions of the turbine blades have been replotted with unit turbine-blade length as a common abscissa. Results of the investigation of the effect of operating temperature on the expected service life of turbine blades of design B at three engine speeds, based on experimentally determined temperatures, are presented in figures 12 to 14.

Centrifugal Stress and Cross-Sectional Area

The characteristics of the turbine blades, as shown in figures 5 to 11, vary sufficiently to warrant individual discussion.

Turbine blade A. - The decrease in cross-sectional area of turbine blade A is almost linear with blade length (fig. 5). The base section is comparatively thick. From the base section, turbine blade A tapers to a relatively thin section at the tip. The slope of the curve for centrifugal-stress distribution is almost constant over its entire length except in the region near the base and tip.

Turbine blade B. - The design characteristics of turbine blade B, as shown in figure 6, are markedly different from those of the other blades studied. The cross-sectional area decreases at a nearly constant rate from the base section to approximately the middle of the blade; from the middle to the tip the area decreases at a slower rate. The decrease in area of both parts of the blade is nearly linear with blade length. As a result of this unique design, the curve representing the distribution of centrifugal stress is dissimilar to those calculated for other unshrouded turbine blades. The stresses in the region near the base of the blade are fairly high. At the middle of the blade, a sudden change in slope occurs and toward the tip of the blade the stresses decrease at a rate very similar to that determined for the other unshrouded blades.

Turbine blade C. - The design characteristics of turbine blade C, which is shrouded, are of especial interest (fig. 7). The reduction of area is not great from the base to the integral shroud segment at the tip. Because of the mass of this shroud segment, it is important that a low centrifugal stress be maintained at the location of change in section. The curve of centrifugal stress differs from those obtained for unshrouded types in that the slope is less along the greater part of the curve. The decrease in both cross-sectional area and in centrifugal stress between base and shroud segment is less than with unshrouded blades of similar dimensions.

Turbine blade D. - The difference between turbine blade D and the others included in the investigation is principally in the quantitative values of its centrifugal-stress and cross-sectional area curves, presented in figure 8. The curve for cross-sectional-area distribution shows a relatively thin base section. The taper decreases parabolically toward the tip. Centrifugal stress at the base is higher than that in any of the other blades.

Turbine blade E. - The length of turbine blade E is approximately equal to that of blade B, but there is no similarity between their respective centrifugal-stress and cross-sectional-area distribution curves. The base section is thick compared with that of turbine blade B and the area curve of figure 9 decreases smoothly to a tip value nearly the same as that of turbine blade B. The curve of centrifugal-stress distribution has a slope similar to the curves for blades A, D, and F. There are no abrupt changes in the slopes of the curves for turbine blade E.

Turbine blade F. - The dimensions of turbine blade F are very similar to those of blade E. A comparison of these two blades shows that blade F has approximately the same base section (fig. 10) as blade E, but does not taper as rapidly. As a result, the centrifugal stress at the base is higher in blade F. The comments made in comparing turbine blade E with the other unshrouded blades are also applicable to turbine blade F.

Comparison of designed stress distribution. - The stress-distribution curves for all the blades plotted against percentage blade length are presented in figure 11. The effect of the change in taper of turbine blade B is clearly shown in this figure. The maximum centrifugal stress is lower in turbine blade C than in any of the other blades.

Temperature-Distribution Effect

Comparisons between rupture stresses for 100- and 1000-hour service lives and the centrifugal stresses present in turbine blade B during operation at engine speeds of 10,000, 11,000, and 11,500 rpm are shown in figures 12, 13, and 14, respectively.

Turbine speed of 10,000 rpm. - The service life of turbine blade B would not be limited by the effect of centrifugal force caused by operation at 10,000 rpm (fig. 12). In the region near the center of the turbine blade, critical because of its high operating temperature, the centrifugal stress is approximately 16,000 pounds per square inch. According to the information obtained from the correlation of blade operating temperature and stress-rupture data, the rupture stress for 100-hour operation is nearly 50,000 pounds per square inch near the center of the blade; the 1000-hour rupture stress is 40,000 pounds per square inch.

Turbine speed of 11,000 rpm. - According to the curves presented in figure 13, failure caused by centrifugal stress alone would occur in turbine blade B before 1000 hours of operation. The rupture stresses as determined for 100-hour operation are sufficiently high, however, to permit safe operation at 11,000 rpm for that period of time, if the effects of stress-inducing factors other than centrifugal force are neglected.

Turbine speed of 11,500 rpm. - At the maximum rated speed of 11,500 rpm, turbine blade B would not withstand operation for 100 hours. Figure 14 shows that the rupture stress would be less at the central part of the turbine blade than the centrifugal stress.

DISCUSSION

The summary and comparison of centrifugal-stress distributions presented in figure 11 show considerable disparity among the designed stresses of the six blade types, although there is similarity in the slopes of the curves except for those of turbine blades B and C. The shape and the slope of the curve for turbine blade C can be explained by the presence of the integral shroud segment at the tip of the blade; the cross section near the tip must be sufficient to withstand the centrifugal force caused by the shroud segment. The comparatively low value of the centrifugal stresses in the lower third (including the base) of turbine blade B is caused by the change in taper near the center of the blade.

The only valid comparisons that can be made among the six turbine-blade types are those based on centrifugal force alone. The rotor radius, design speed, blade length, and blade shape of turbine blade C result in a stress distribution that makes this blade most resistant to failure on the basis of stress rupture.

Turbine blade D is subjected to consistently high stresses, relative to the values that occur in the other turbine blades studied.

The centrifugal stresses in turbine blade F are higher than those determined for blade E. It is known, however, that the operating temperature in turbine blade F, the second-stage blade, is lower than in turbine blade E, the first-stage blade. It is therefore probable that the effect of higher centrifugal stresses in the second-stage turbine blade is offset to some extent by the higher rupture strength.

The comparisons between centrifugal stresses and predicted rupture stresses shown in figures 12 to 14 for turbine blade B at different operating speeds are of especial interest. At a speed of 10,000 rpm, near the lower limit for cruising, the turbine blade would operate for an indefinite period of time without failure from stress rupture caused by centrifugal force, as shown in figure 12. Figure 13, however, indicates that the centrifugal stresses in the central portion of turbine blade B would exceed the rupture strength for 1000-hour operation at 11,000 rpm. The curves shown in figure 14 show that the centrifugal-stress distribution is sufficiently high to limit safe operation at 11,500 rpm to less than 100 hours.

The operating temperatures of the other turbine blades are probably similar to that of turbine blade B. During service operation, the tail-pipe gas temperatures of the various units are approximately the same. The operating temperature of turbine blade E may be higher, however, than those of the other turbine blades because of its location in the first stage of a two-stage turbine.

The importance of temperature distribution can be exemplified by the fact that service failures have occurred in the region predicted by the calculations made for turbine blade B. The region where failures have occurred in such a turbine blade is indicated in figure 15. The appearance of such failures indicates that the rupture begins at the trailing edge within the rather narrow limits shown in the figure.

SIGNIFICANCE OF RESULTS

The analytical investigation of centrifugal-stress and cross-sectional-area distributions in six types of gas-turbine blade indicates that changes in the current design of turbine units to effect a reduction in centrifugal stresses and operating temperatures of the blades would be advantageous. The fact that service failures have occurred in the region indicated as probable by the analysis tends to confirm the assumption that the limitation of centrifugal stresses by operating temperatures is an important factor in considerations involving service life of turbine blades.

Flight Propulsion Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, December 5, 1947.

REFERENCE

1. Cross, Howard C., and Simmons, Ward F.: Prog. Rep. on Heat-Resisting Metals for Gas Turbine Parts (N-102). OSRD No. 4717, NDRC, OSRD, War Metallurgy Div., Feb. 20, 1945. (Abs. Bib. Sci. Ind. Reps., vol. 1, no. 18, May 10, 1946, p. 1024, PB 18484.)

865

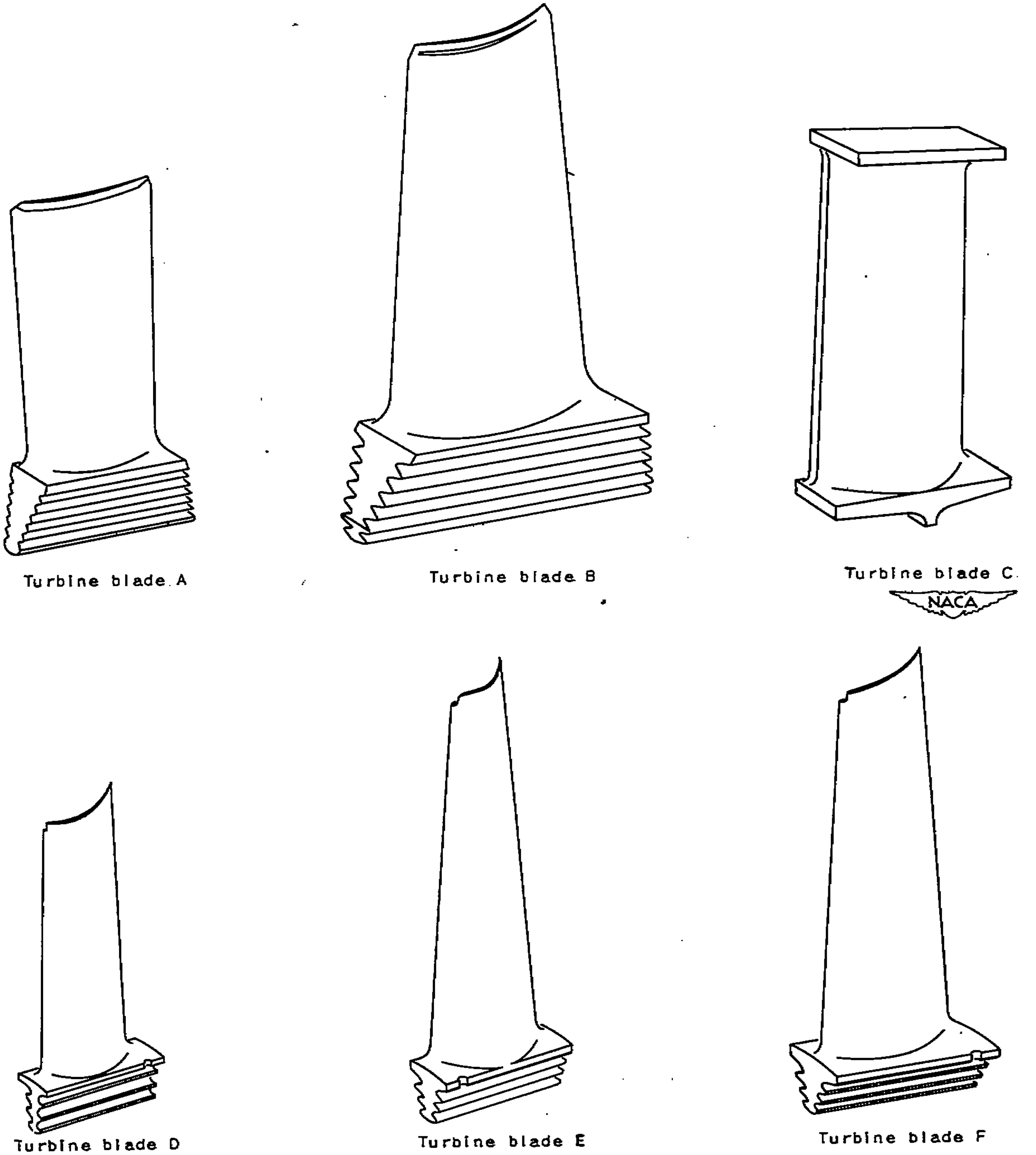

Figure 1. - Three-quarter views of six turbine-blade designs selected for analytical investigation of centrifugal-stress distribution during full-speed operation.

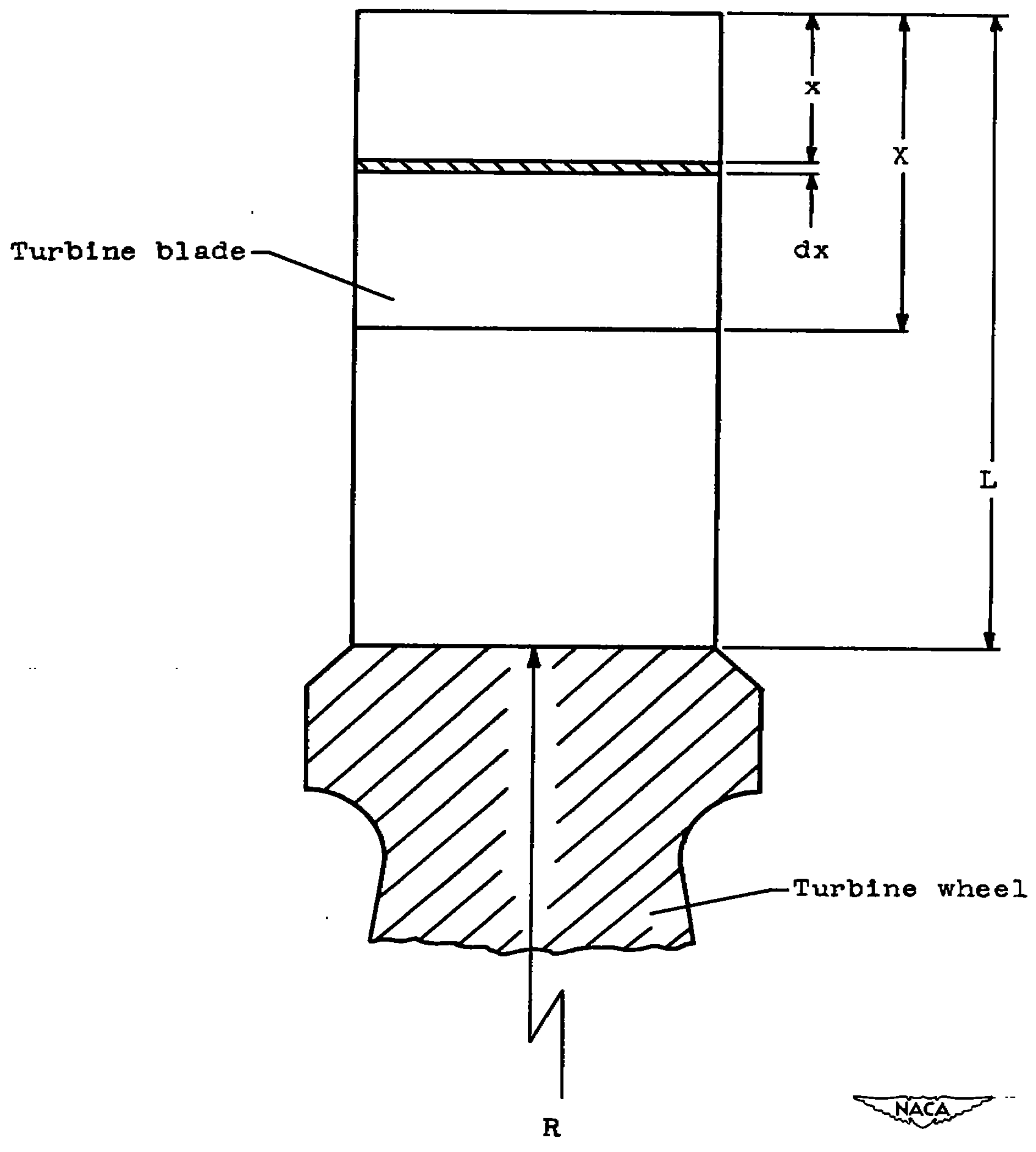

Figure 2. - Symbols used in derivation of centrifugal force formula (equation (1)).

865

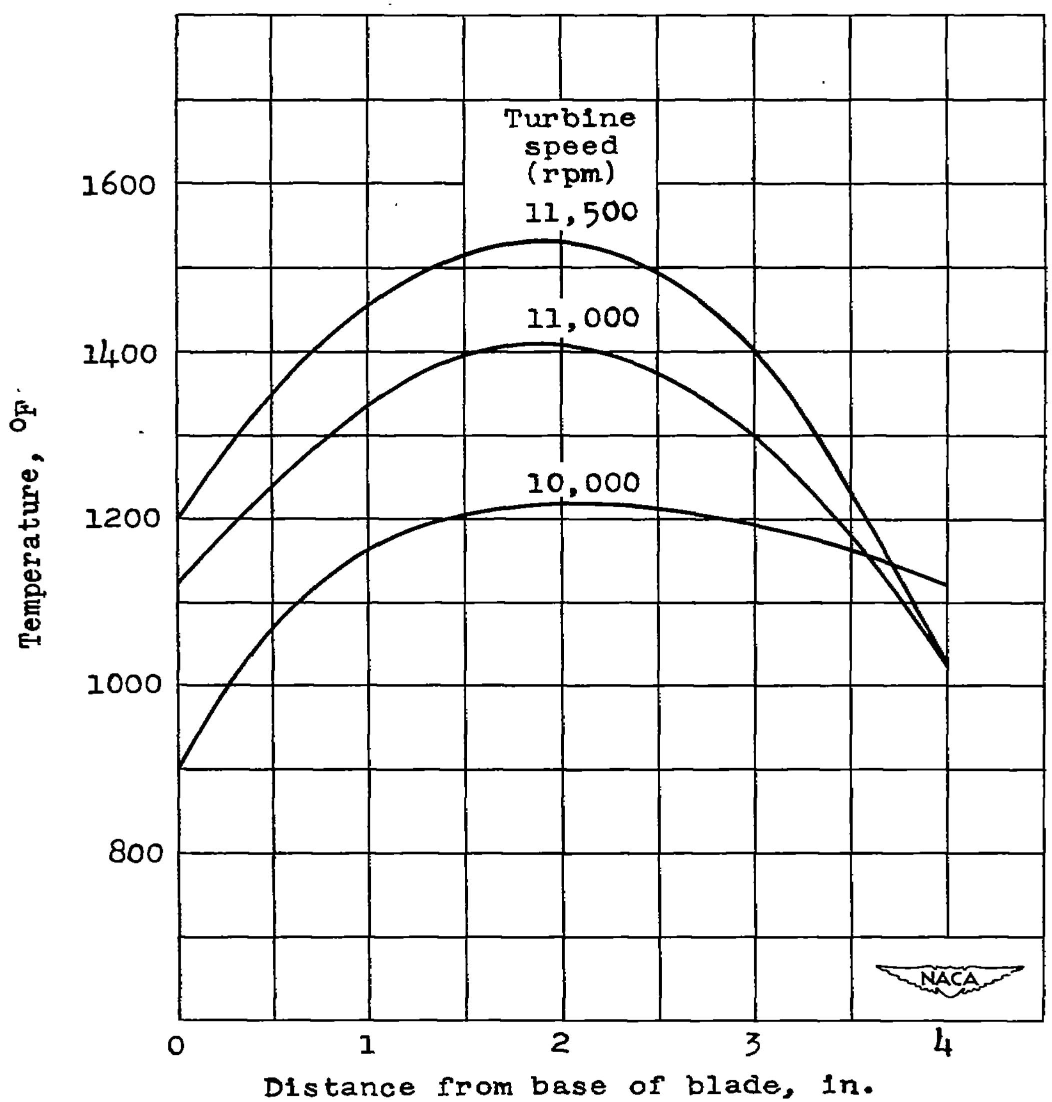

Figure 3. – Variation of temperature in turbine blade B during service operation at several turbine speeds.

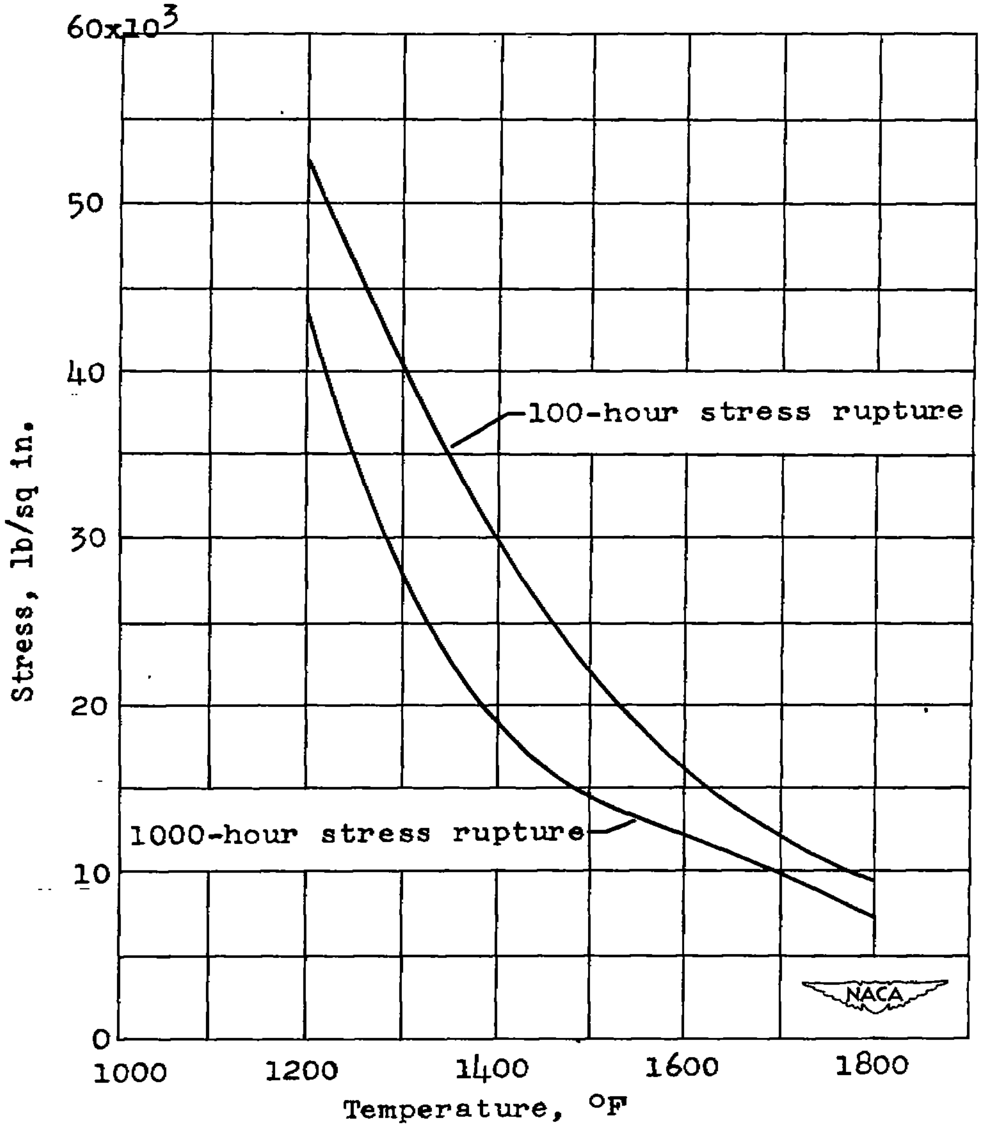

Figure 4. – Variation of stress with temperature for Vitallium in 100-hour and 1000-hour stress-rupture tests.

865

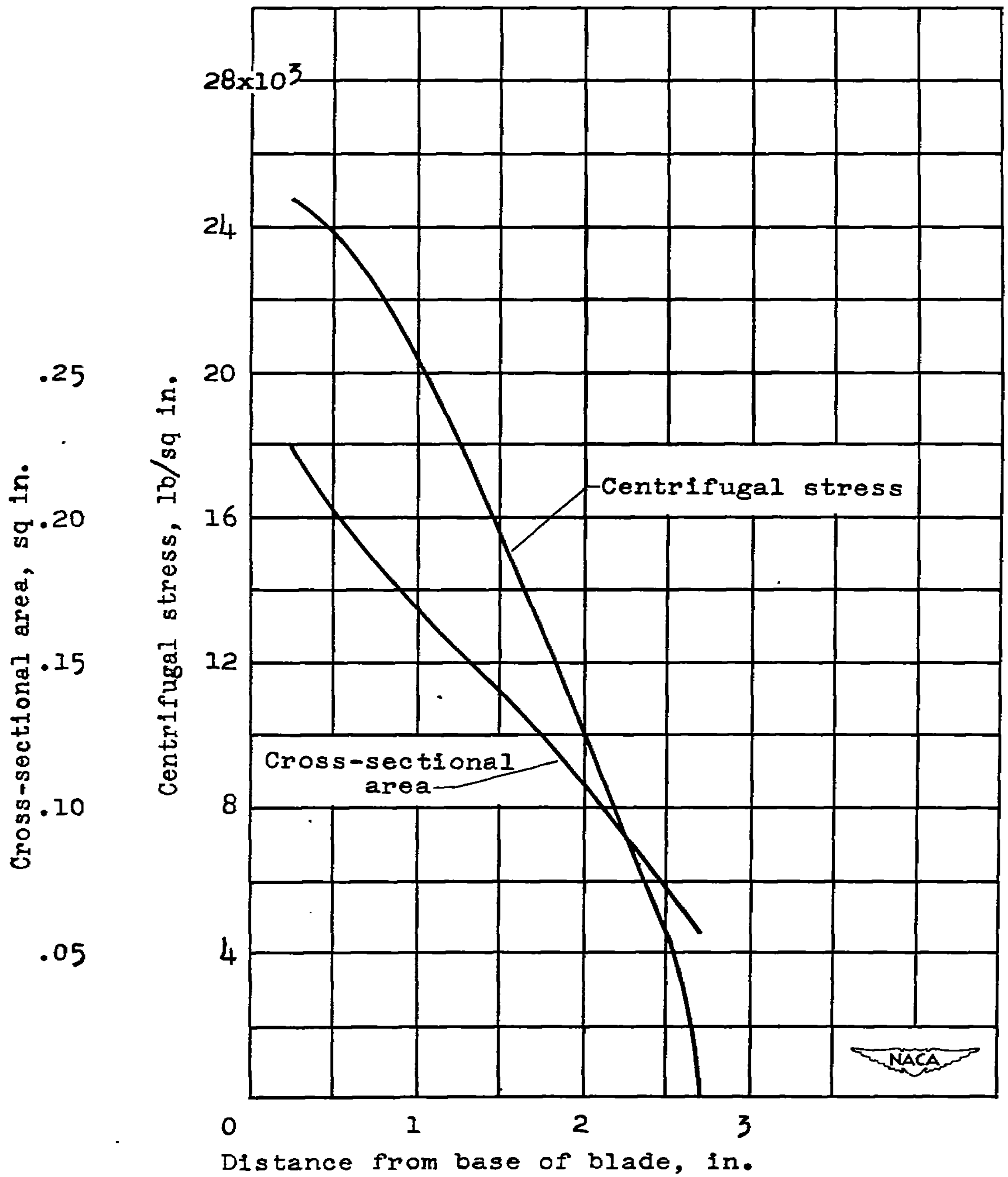

Figure 5. – Variation in centrifugal stress and cross-sectional area in turbine blade A. Turbine speed, 16,500 rpm; radius of turbine rotor, 5.75 inches.

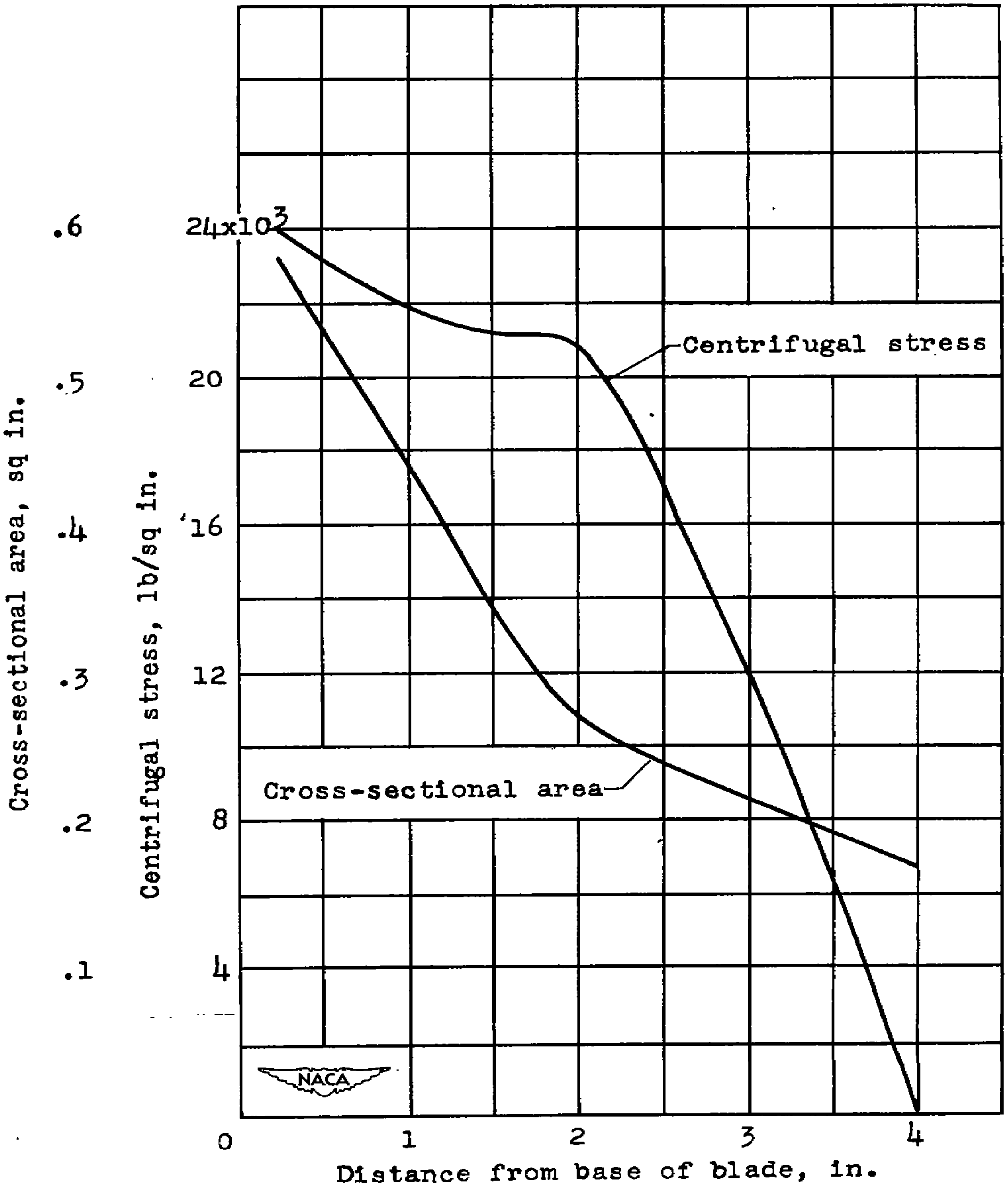

Figure 6. - Variation in centrifugal stress and cross-sectional area in turbine blade B. Turbine speed, 11,500 rpm; radius of turbine rotor, 8.99 inches.

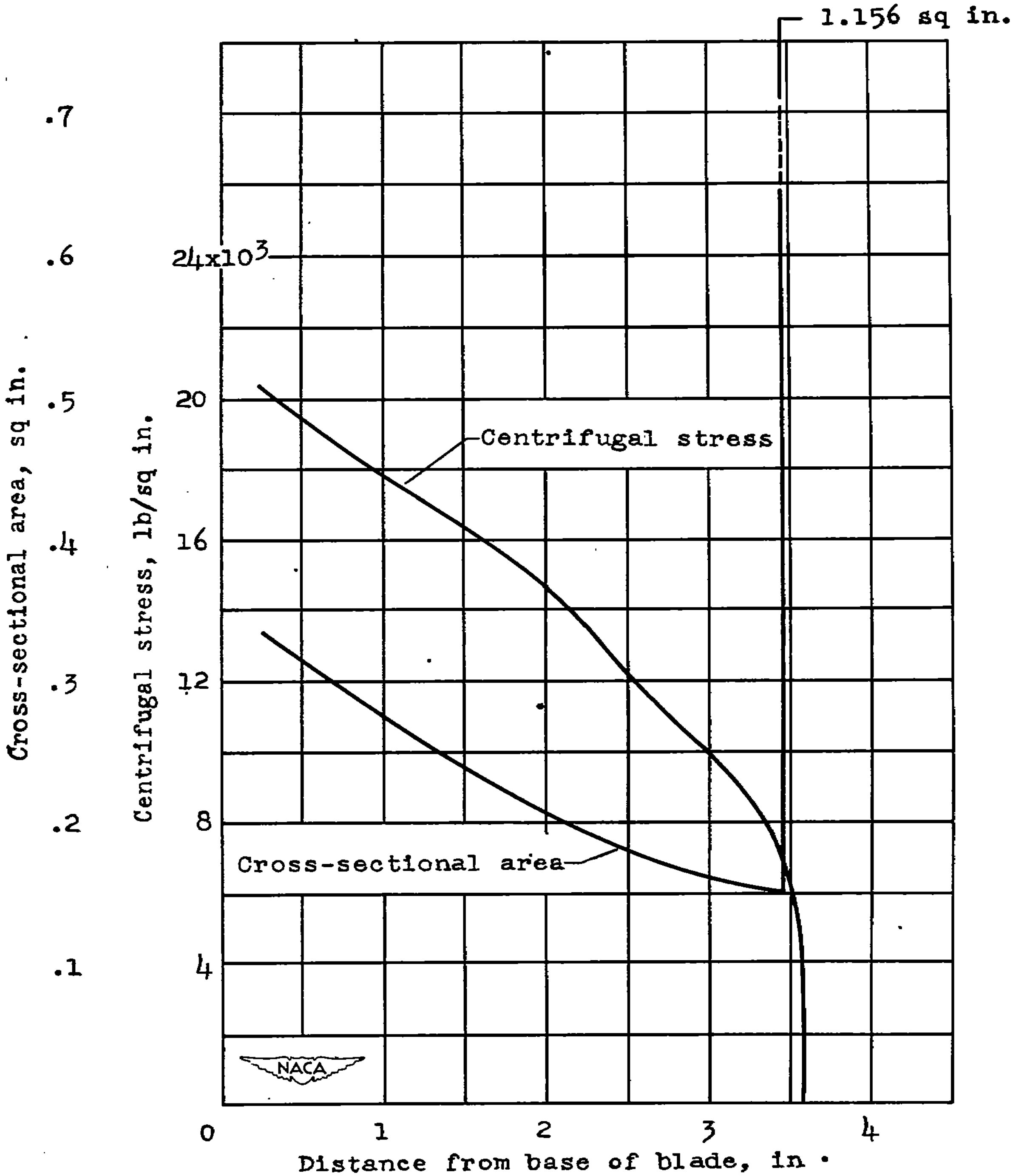

Figure 7. - Variation in centrifugal stress and cross-sectional area in turbine blade C. Turbine speed, 7600 rpm; radius of turbine rotor, 13.47 inches.

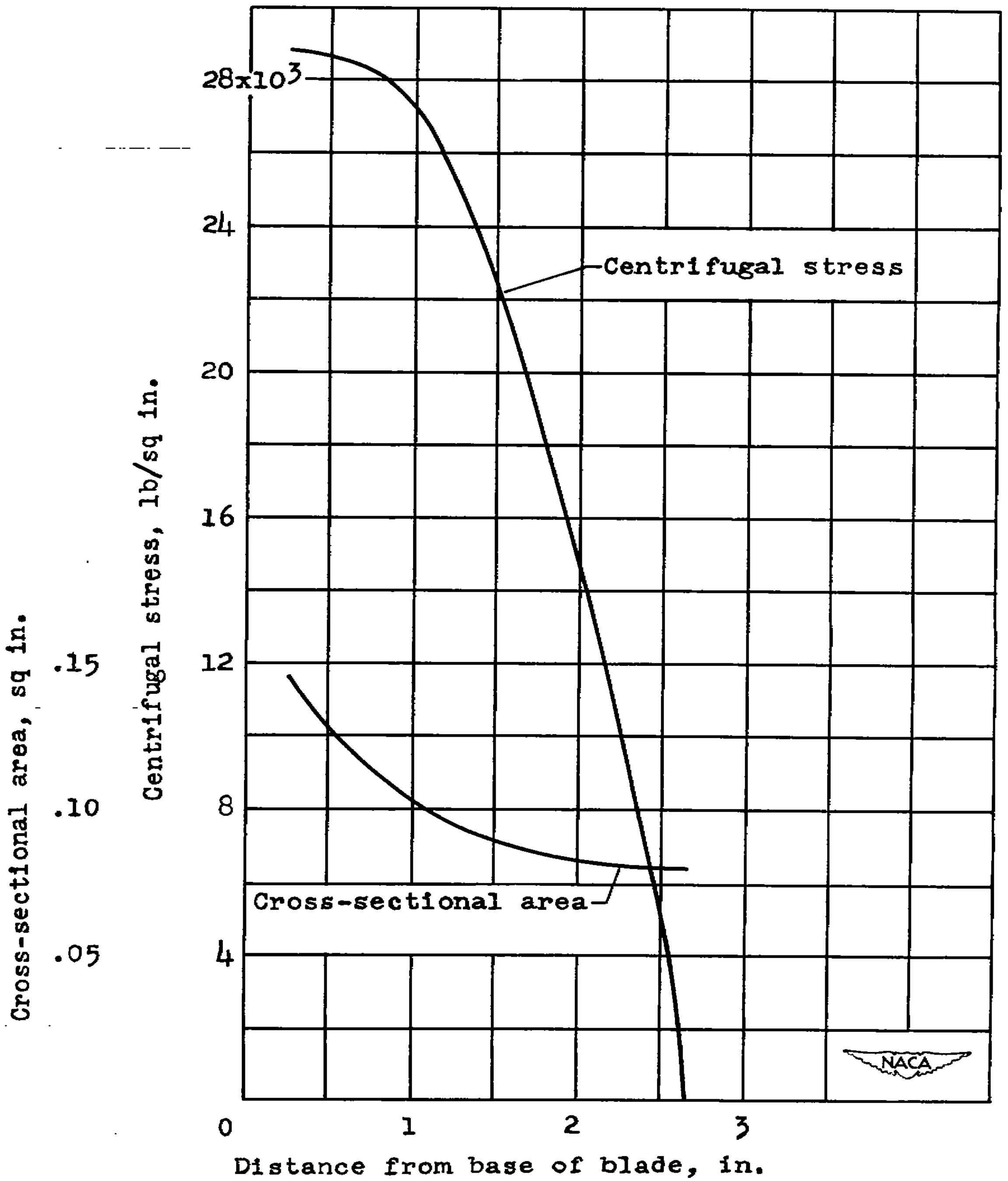

Figure 8. - Variation in centrifugal stress and cross-sectional area in turbine blade D. Turbine speed, 16,500 rpm; radius of turbine rotor, 5.94 inches.

865

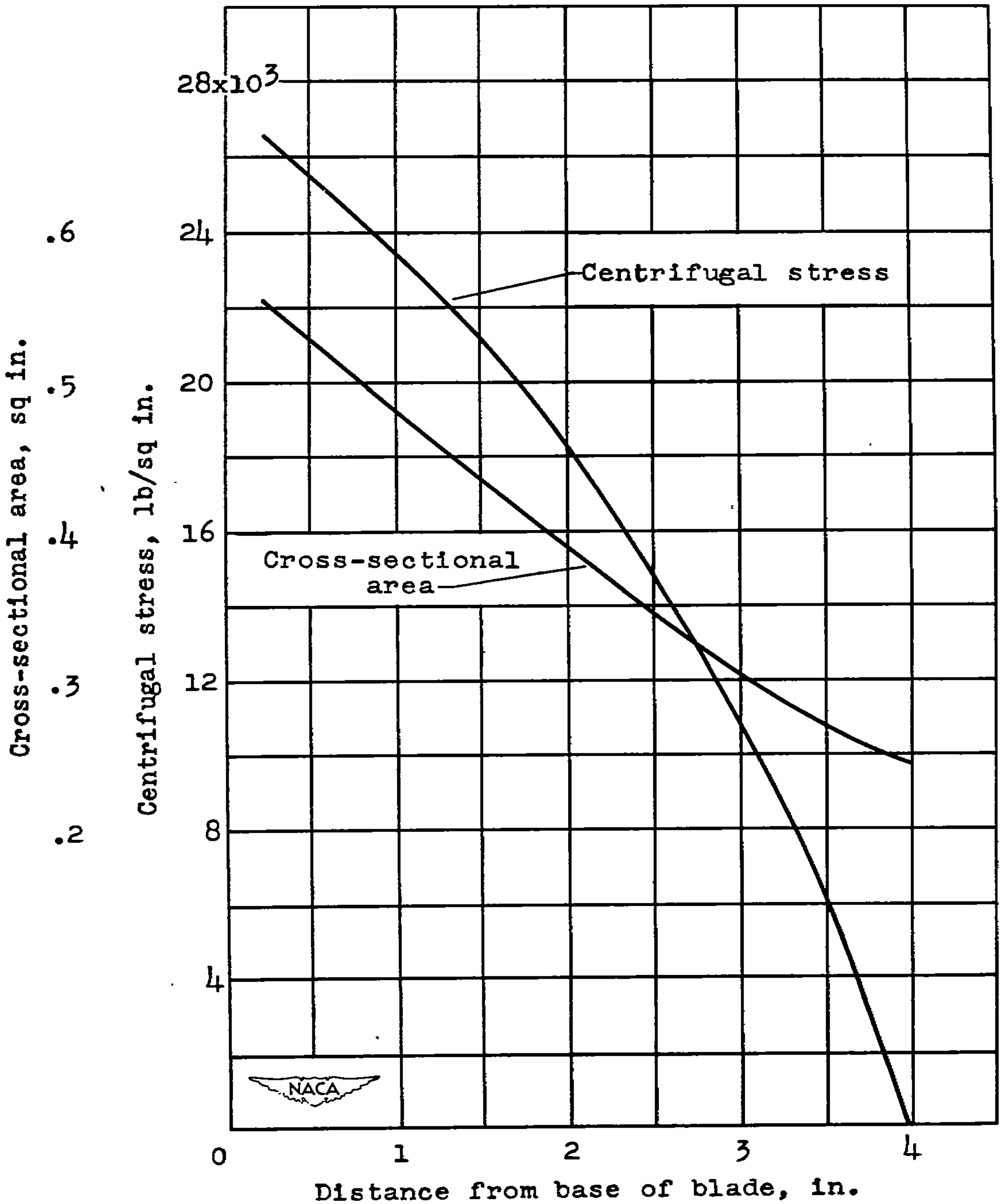

Figure 9. - Variation in centrifugal stress and cross-sectional area in turbine blade E. Turbine speed, 12,000 rpm; radius of turbine rotor, 6.50 inches.

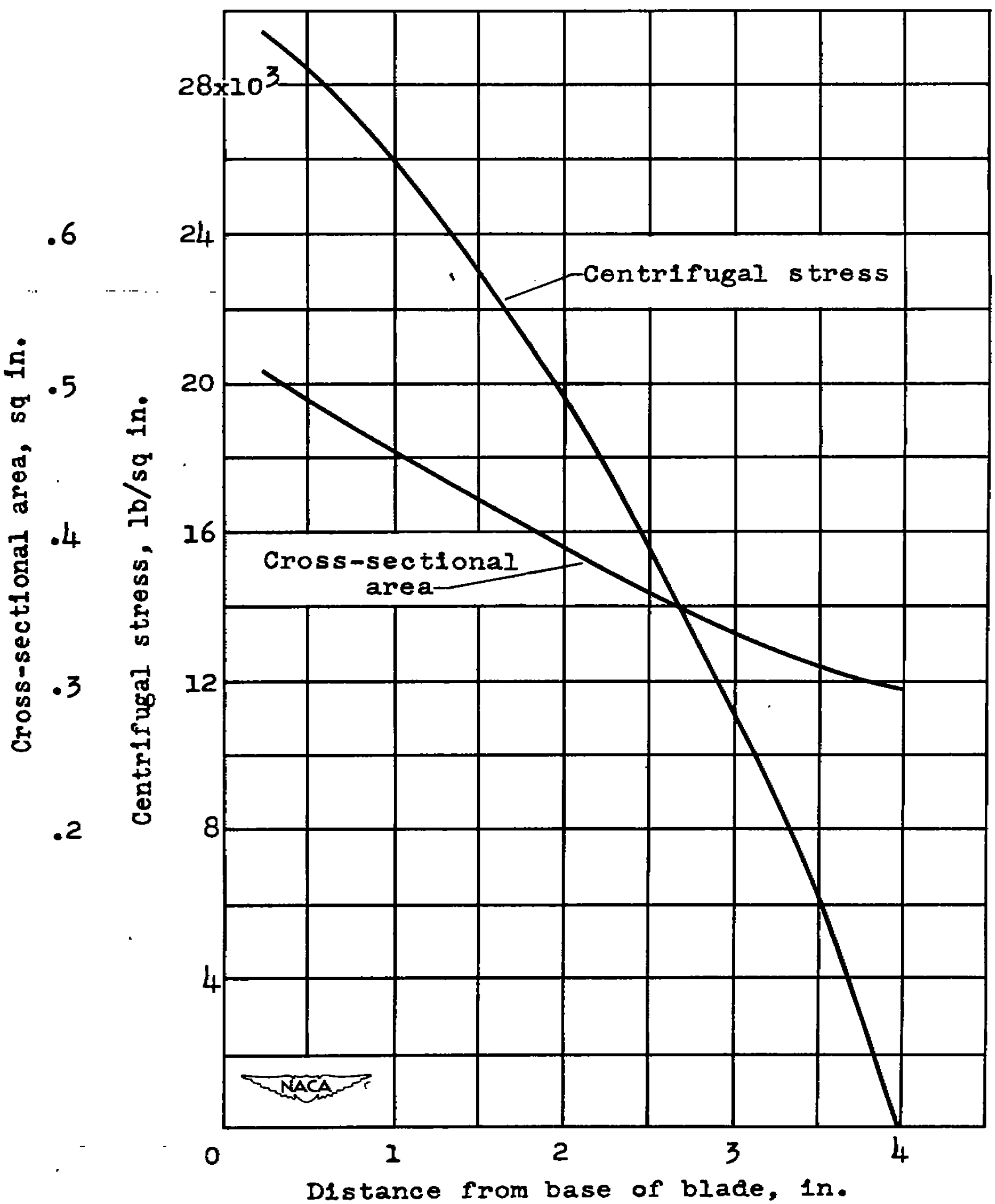

Figure 10. – Variation in centrifugal stress and cross-sectional area in turbine blade F. Turbine speed, 12,000 rpm; radius of turbine rotor, 6.50 inches.

865

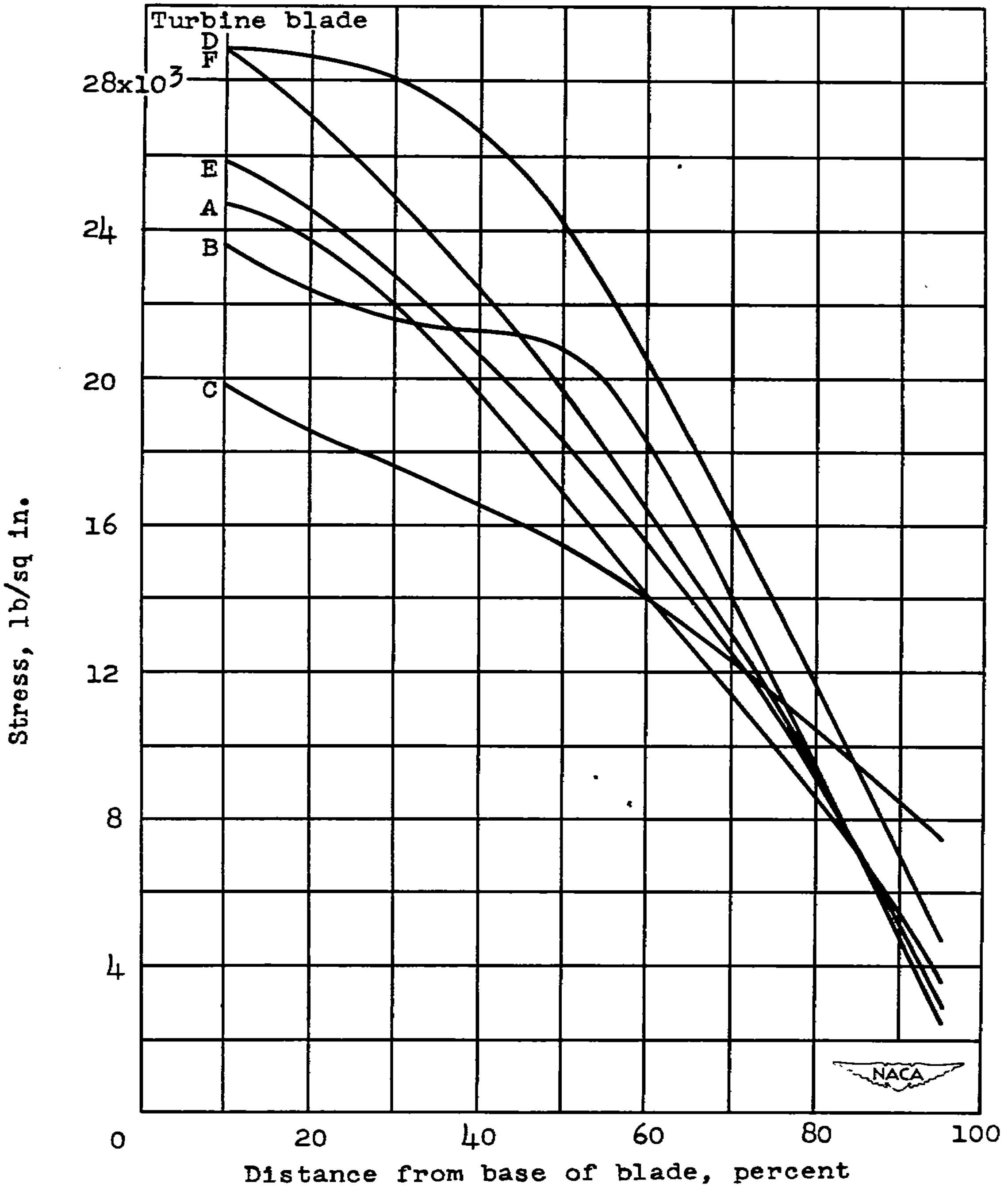

Figure 11. - Comparison of centrifugal stress distributions present in turbine blades of several designs at full-speed operation.

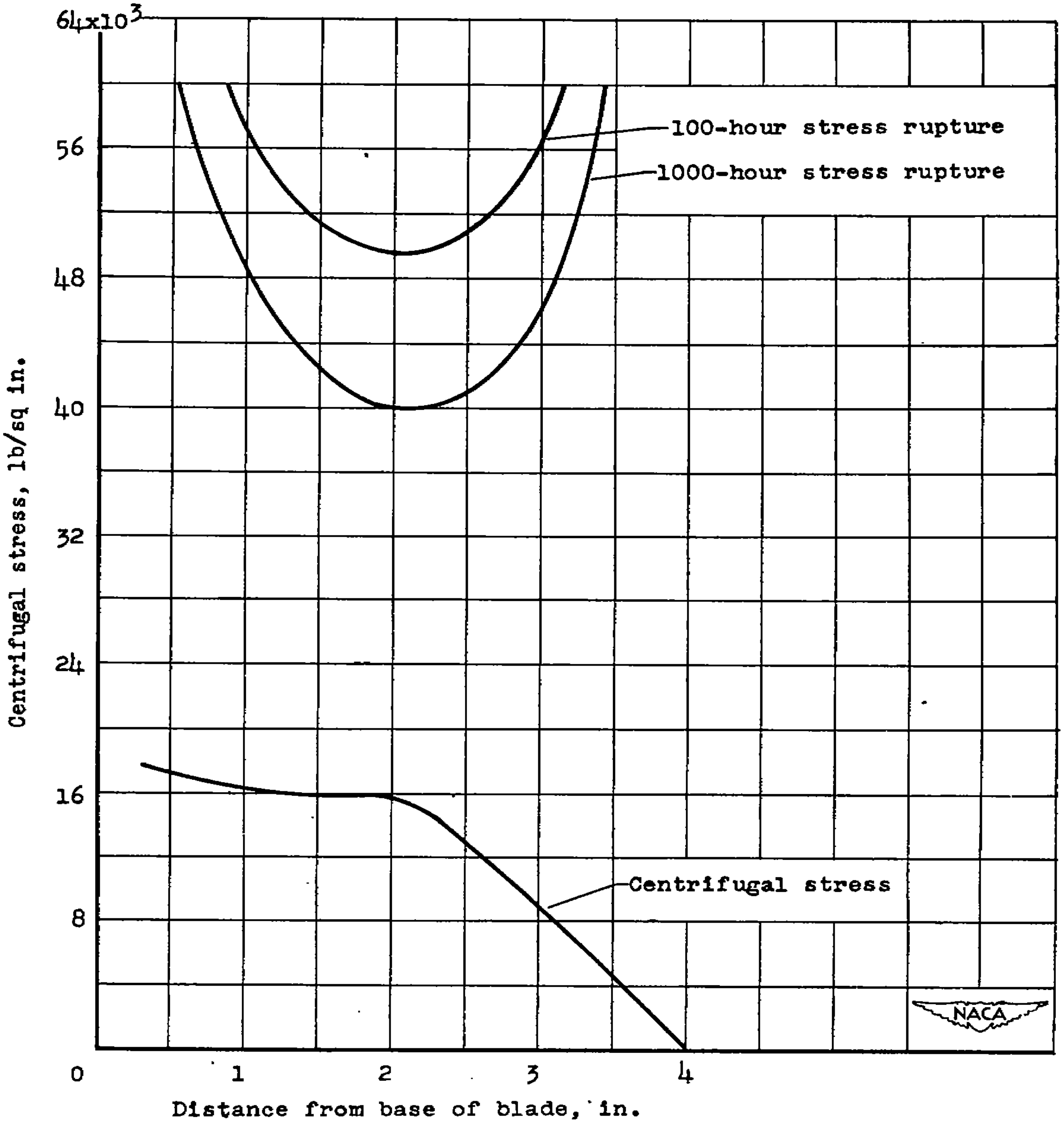

Figure 12. - Comparison between calculated centrifugal stresses and allowable stresses as computed from stress-rupture data and operating-temperature data for turbine blade B. Turbine speed, 10,000 rpm.

865

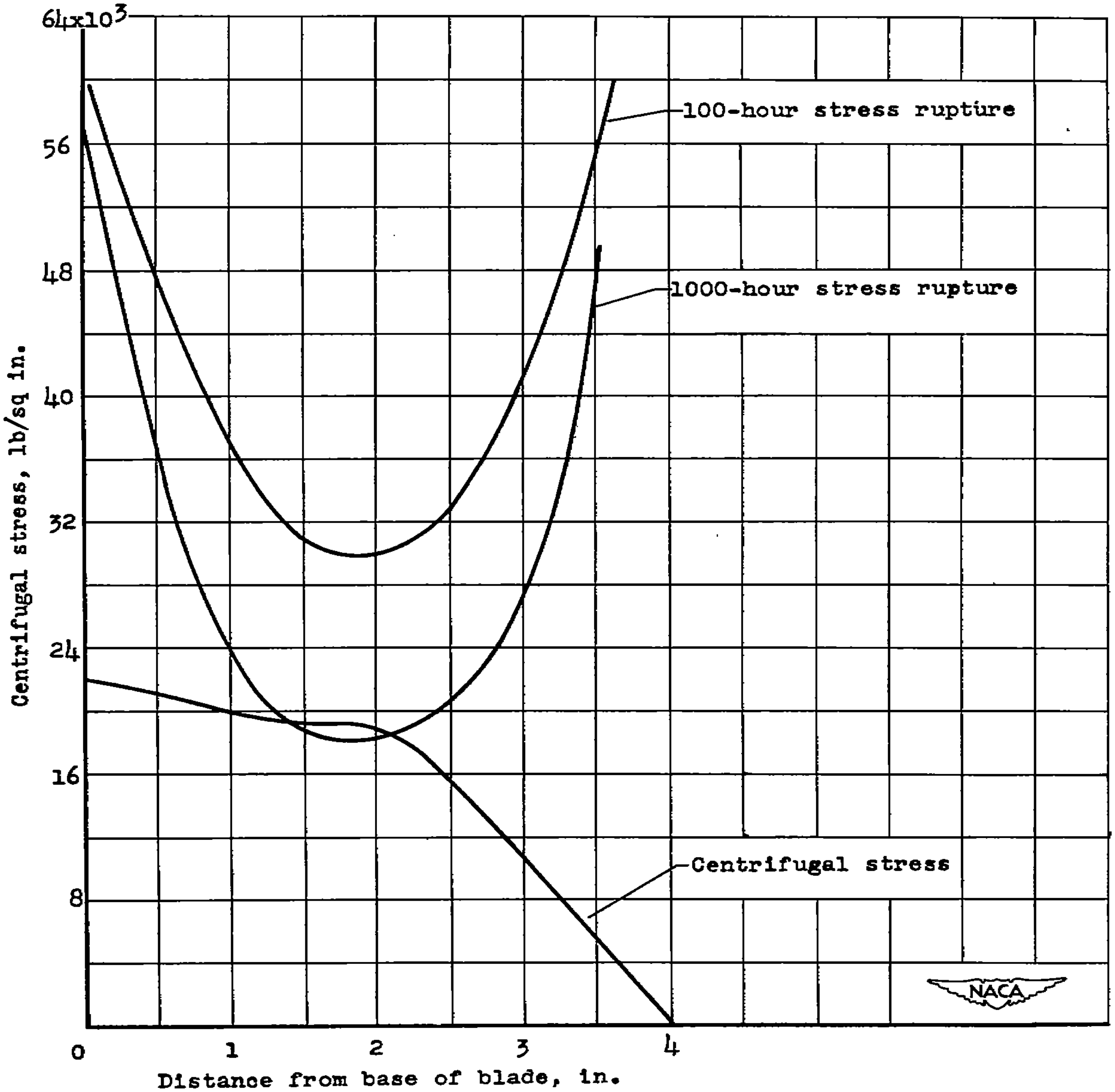

Figure 13. - Comparison between calculated centrifugal stresses and allowable stresses as computed from stress-rupture data and operating-temperature data for turbine blade B. Turbine speed, 11,000 rpm.

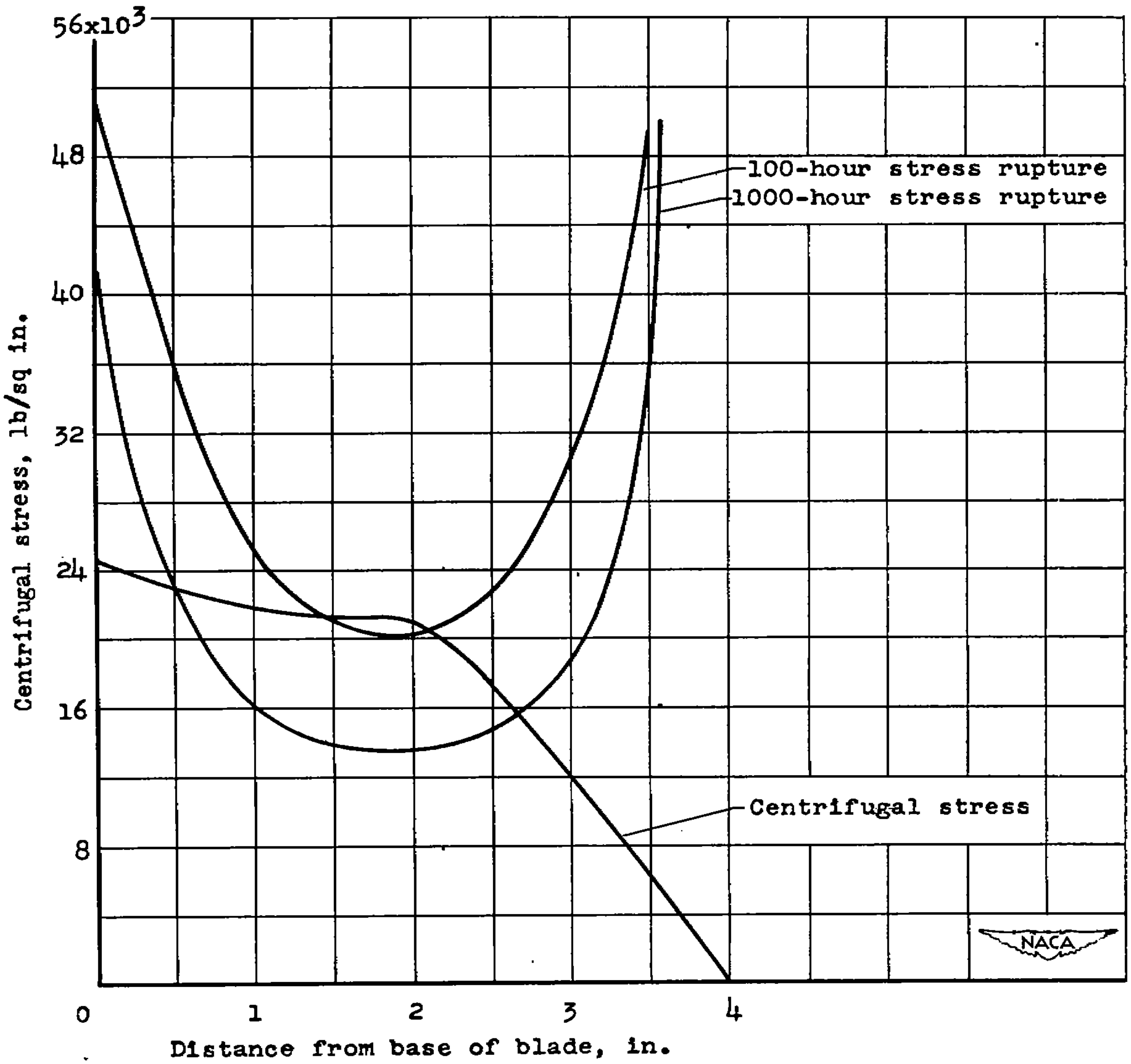

Figure 14. - Comparison between calculated centrifugal stresses and allowable stresses as computed from stress-rupture data and operating-temperature data for turbine blade B. Turbine speed, 11,500 rpm.

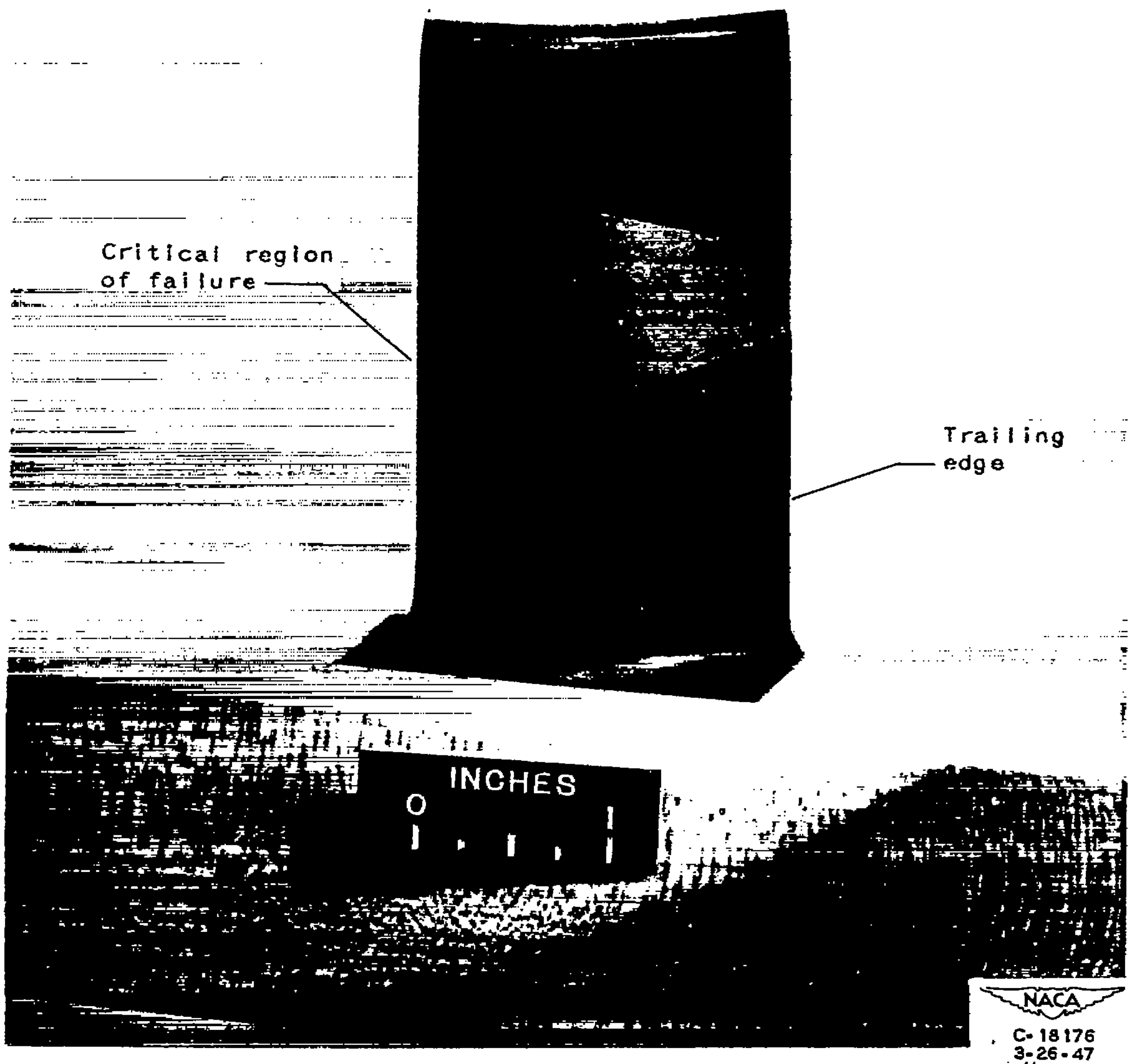

Figure 15. – Critical region of failure in turbine blade B, based on records of service failures.